

1994

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Zanelli, R. and Favrat, D., "Experimental Investigation of a Hermetic Scroll Expander-Generator" (1994). *International Compressor Engineering Conference*. Paper 1021.

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# EXPERIMENTAL INVESTIGATION OF A HERMETIC SCROLL EXPANDER-GENERATOR

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## ABSTRACT

This paper describes a 1 to 3.5 kWe hermetic scroll expander-generator modified from a standard hermetic compressor, an organic Rankine cycle test facility built to test expanders up to 10 kWe and a set of experimental results using HCF 134a in the dry vapor domain. Peak overall isentropic efficiencies in the range of 63 to 65% for speeds of rotation varying between 2400 and 3600 rpm are reported. Performance is fairly constant in the range of pressure ratios considered ( $2.4 < PR < 4.0$ ).

## NOMENCLATURE

$\dot{E}_{\text{electric}}$	[W]	electric power
$F_f$	[-]	filling factor
$\Delta h_s$	[J/kg]	isentropic enthalpy drop per unit of refrigerant mass
$\dot{m}$	[kg/s]	refrigerant mass flow rate
$S_v$	[m <sup>3</sup> /rev]	suction volume
$v_{\text{in}}$	[m <sup>3</sup> /kg]	inlet specific volume
$\eta_s$	[-]	overall isentropic efficiency
$v$	[Hz]	shaft speed rotation

## INTRODUCTION

New machinery developments are needed in order to give satisfactory answers to current concerns over environmental issues related to energy conversion and use as well as to the long term objective of preserving the Earth's resources. Small scale organic Rankine cycles (ORC) for electro-thermo-solar conversion, waste heat recovery or as topping cycles of boilers or absorption (refrigeration) systems could contribute to this objective but often suffer from a lack of efficient and reliable expanders or turbines.

In the small to medium range of electric power and with hermetic units able to comply with present environmental concerns, scroll expanders present definite advantages over other potential volumetric machines. In addition, the reduced number of parts, the ability to cope with two phase working fluids and the inherent reliability proven in compressor mode are major reasons for its wider use in energy recovery. The fast growing mass production of such units as compressors is likely to also contribute to a lowering of the production costs for expander applications.

## CONVERSION OF A SCROLL COMPRESSOR UNIT FOR EXPANDER OPERATION

As is the case for all rotary volumetric principles, backward motion is inherently possible and, therefore, the conversion of most types of scroll compressors to expander mode can be envisaged. Our initial approach was to modify a hermetic unit from the same family of a compressor previously tested with HCFC22 in an air to water heat pump [1]. The main features of the original compressor unit are: a suction volume of 78.41 cc/rev, a built-in volume ratio of 2.44, a nominal electric power of 3.75 kWe (3 phased asynchronous 2 pole motor) and a nominal rotation speed of 2880 rpm [2]. In compressor mode, the unit is discharged cooled with the higher pressure refrigerant in the shell and has a simple differential pressure oil lubricating system, a fixed orbiting radius and, a self-adjusting back pressure mechanism coupled with a hydrostatic oil thrust bearing. The original built-in suction check valve of the compressor unit was removed.

### Refrigerant flow:

Although several working fluids, as for example propane or butane, could be considered for the desired applications, HFC 134a was chosen for ease of use, non flammability and global environmental acceptance. As shown in figure 1, high pressure vapor is directly fed through the central port without transiting through the shell. A side hole in the inlet pipe maintains the high pressure in the shell. The low pressure refrigerant-oil mixture is then discharged through an oil separator upstream of the condenser.

### Oil flow:

Mobil polyol-ester EAL Artic 68 oil is used. After being separated, oil is pumped back to the feeding pockets of the hydrostatic axial thrust bearing. The oil excess falls to the bottom of the shell where it is sucked up by the central hollow shaft in order to lubricate other bearing parts and the working chambers. The oil flow is controlled by a level sensor probe. No oil is fed to the inlet pipe.

### Electrical system:

The 3 phased asynchronous generator is connected through a digital wattmeter to a set of capacitors, an adjustable voltage transformer and a fixed resistive load. This combination was adjusted for each load case in order to generate the minimum reactive power needed for the proper functioning of the unit in generator mode.

## ORGANIC RANKINE CYCLE TEST FACILITY

The cycle presently operates between a maximum heat source temperature of 80 °C (hot water) and a minimum heat sink temperature of 5 °C (industrial cooling water). The facility is equipped with a plate evaporator, a plate superheater and a plate condenser, as well as a membrane liquid refrigerant feed pump driven by an inverter. A refrigerant liquid bypass upstream of the evaporator enables an easy adjustment of the expander inlet conditions in the superheated as well as in the wet domains. The latter is particularly useful when assessing control strategies for the ORC system with a varying heat supply such as is found, for example, in solar systems. A complementary fluid tank branched in parallel with the main flow circuit allows the adjustment of the refrigerant charge when the cycle is at rest. With 60 kW available at the evaporator, turbines with a power of up to 10 kWe could be tested at a maximum pressure of 2500 kPa.

## INSTRUMENTATION

The expander is instrumented with inlet and discharge pressure transducers, inlet and discharge temperature sensors (type K thermocouples) and a high precision digital wattmeter. The shaft rotation speed is recorded through a dynamic pressure sensor located at the pipe inlet.

The ORC facility instrumentation includes two coriolis flow meters and basic temperature and pressure sensors which permit the determination of the energy balance for each component and the management of the plant. Each sensor is individually and carefully calibrated.

Data acquisition and controls are done with a Hewlett Packard 3852a data acquisition system and with a National Instrument analog output board driven by LabVIEW software on a Macintosh PC.

## RESULTS AND DISCUSSION

Present results cover primarily superheated conditions, with a fixed inlet temperature of 70°C and a fixed discharge pressure of 450 kPa for four speeds of rotation (40, 50, 60 and 70 Hz). Continuous signal display is used to ensure steady state conditions before recording test data. Processing of the data is based on fluid thermodynamic properties calculated with a Lee-Kesler equations of state software package [3]. Definitions used are as follows:

$$\text{Overall isentropic efficiency: } \eta_s = \frac{\dot{E}_{\text{electric}}}{\dot{m} \cdot \Delta h_s}$$
$$\text{Filling factor : } F_f = \frac{\dot{m}}{v \cdot S_v / v_{in}}$$

The results plotted in figures 4 and 5 show the most significant performance values. Power for auxiliaries (oil and refrigerant pumps) is not considered. The overall isentropic efficiency is given as a function of the ratio of the measured pressure ratio to the built-in adapted pressure ratio. The latter is calculated on the basis of an isentropic expansion corresponding to the built-in volume ratio. The overall isentropic efficiency is fairly flat over most of the pressure ratio range considered (pressure ratio varying between 100 and 160% of the built-in pressure ratio) and for speeds of rotation close to the nominal value (40 to 60 Hz). Lower and more scattered results observed at 70 Hz can be partly explained by the unsatisfactory lubrication conditions at such high speed. Peak efficiencies approaching 65% have been measured at the nominal speed of 50 Hz (3000 rpm) and a pressure ratio of 3.2. As expected, peak efficiencies occur at increasing pressure ratios as the speed of rotation increases with a resultant relative decrease in mechanical losses.

The filling factor exhibits a fairly constant behaviour at a given speed of rotation over the range of pressure ratios tested. The decrease of the filling factor with increasing rotation speeds is consistent with the relative decrease of the internal clearance losses and the increase of the inlet throttling losses.

The electric power curves shown in figure 6 exhibit a close to linear behaviour as a function of the pressure ratio for the three best speeds of rotation identified. On the basis of these results and the ones

above and provided that the lubrication system is improved, prospects exist for the use with satisfactory performance of such expanders over an even wider range of rotation speed than previously explored .

Finally, the oil mass flow is not accurately measured but can be estimated from the volumetric pump rating to be of the order of 8% of the mass flow of refrigerant for all test points. Because of the relative warming effect of the oil in the working chambers, internal isentropic efficiencies ( enthalpy drop over isentropic enthalpy drop) do not fully reflect the effective internal performance of the machine. For the best range of speed, they vary between 75 and 58 % for effective pressure ratios varying from 100% to 160% of the adapted pressure ratio. A detailed simulation model is presently under way which will, among other factors, account for these internal heat transfer effects and allow an accurate determination of the associated polytropic efficiencies.

### CONCLUSION

The successful conversion of a hermetic scroll compressor into a scroll expander-generator followed by promising performance results opens the way to a new class of modular ORC units in the small power range. HFC 134a proved to be a suitable candidate as a working fluid for such a purpose. Compact plate evaporators and condensers successfully used in the dedicated test rig built for these tests confirmed their suitability for the contemplated applications. Future work includes expansion in the wet region (a condensing expander), detailed simulation modeling, the test of a higher capacity unit with a low shell pressure and the coupling with a higher temperature ORC stage.

### ACKNOWLEDGMENTS

The authors wish to thank the Swiss Federal Office of Energy for their financial support.

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FIGURES

Figure 1: Modified scroll expander

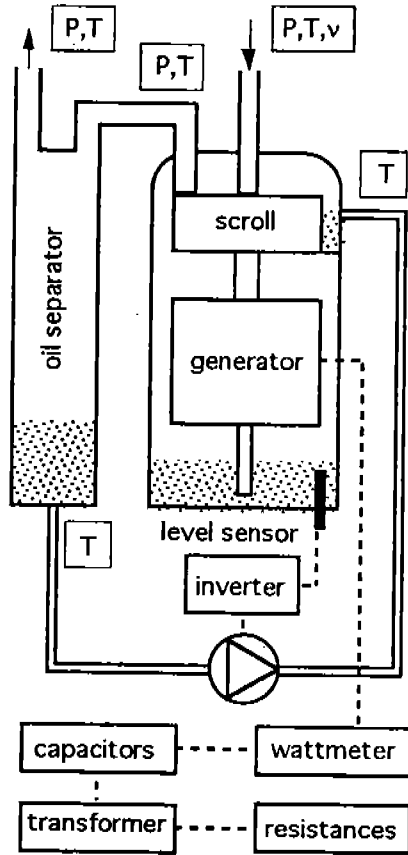


Figure 2: Diagram of the ORC system

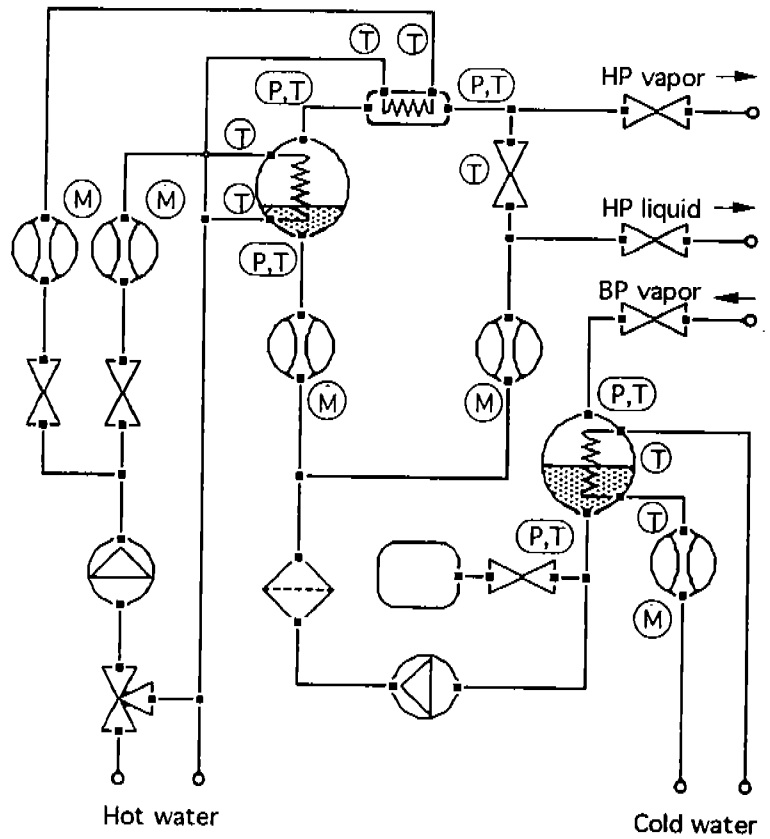


Figure 3: Photograph of the ORC test facility

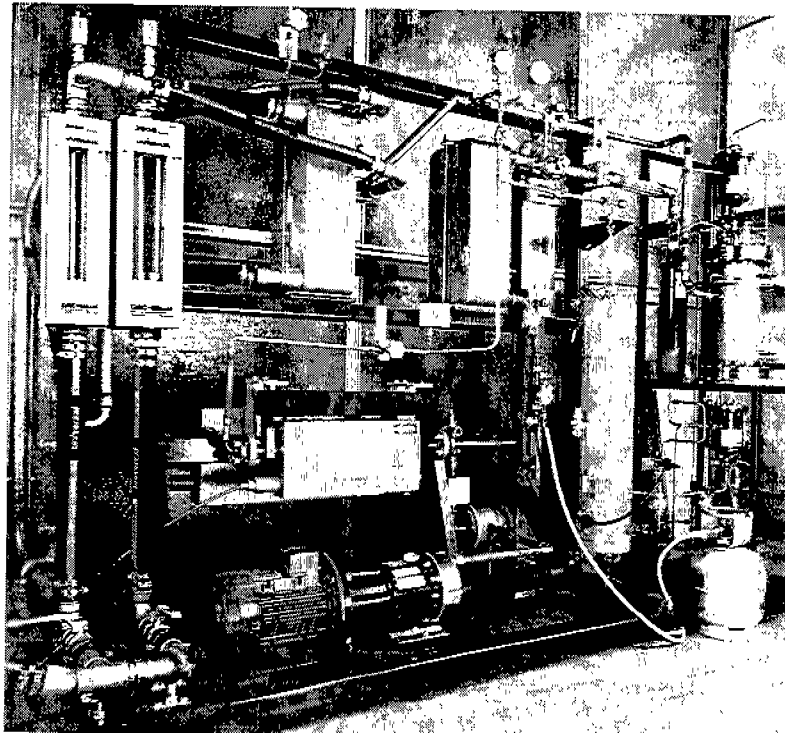


Figure 4: Overall Isentropic Efficiencies

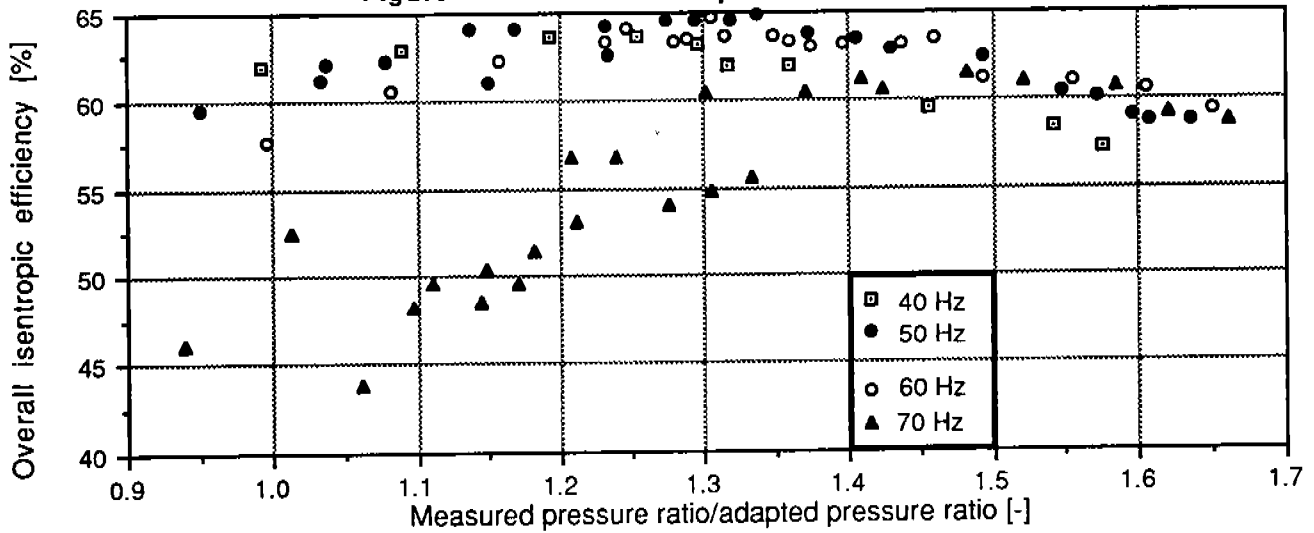


Figure 5: Filling Factor

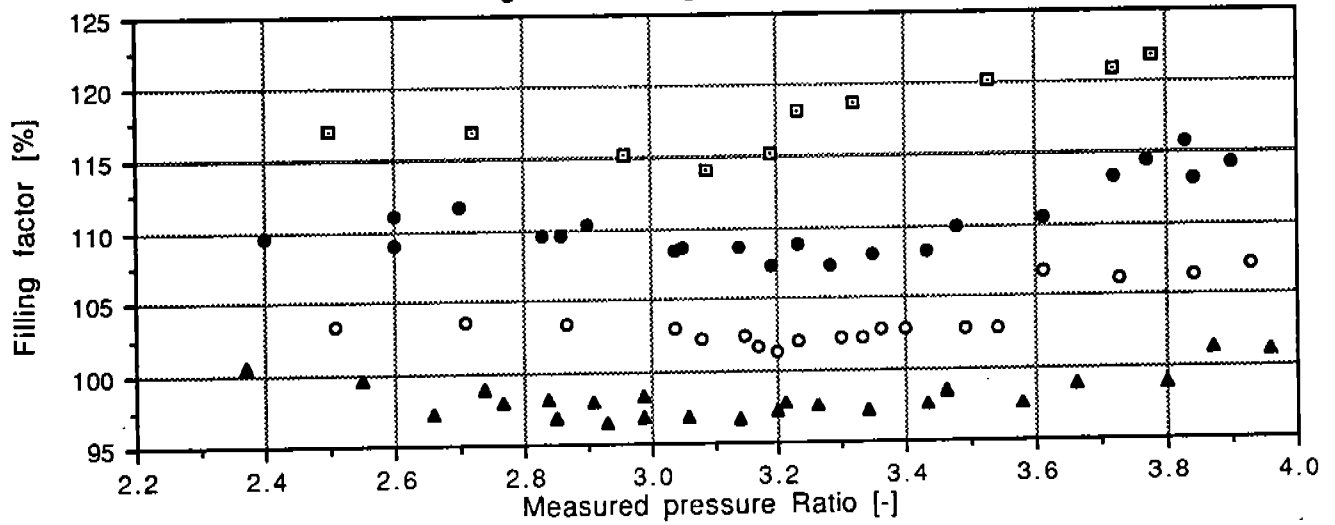


Figure 6: Electrical power

